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Island Concept EVT

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ABSTRACT

This paper presents an all-wheel-drive (AWD) hybrid electric vehicle (HEV) design approach for extreme off-road dynamic performances focusing primarily on the powertrain design, modeling, simulation, and performance analysis. Since this project focuses on a military-type application, the powertrain is designed to enhance crew survivability and provide several different modes of limp operation by utilizing a new vehicle topology –herein referred to as the island topology. This topology consists of designing the vehicle such that the powertrain and other equipment and subsystems surround the crew compartment to provide a high level of protection against munitions and other harmful ordnance. Thus, in the event of an external shield penetration, the crew compartment remains protected by the surrounding equipment –which serves as a secondary shield. The powertrain system principally consists of two internal combustion engines which are coupled to three electric machines using two planetary gear sets, and also includes other transmission elements. A detailed design of the powertrain is presented, which considers the vehicle layout, space claim issues, and extreme operating conditions.

Next in the paper, the mathematical model of the powertrain is outlined. The powertrain model is quasi-static, meaning that it neglects the high-frequency dynamics due to gear elasticity. The model however, takes into account the inertia of the various components, as well as accounting for the parasitic losses due to friction. Finally, the powertrain model is implemented in a simulator program that provides an all-encompassing tool that allows for a complete analysis of the energy flow within the powertrain components. With this, the powertrain control strategy can be formulated with the objective of minimizing fuel consumption and / or maximizing performance using a trade-off approach.

INTRODUCTION

The Island Concept EVT design is being jointly generated by the Center for Automotive Research (CAR) at the Ohio State University and the US Army Tank-Automotive Research, Development, and Engineering Center (TARDEC) as part of a collaborative research project that is currently underway to accomplish an overall design concept of a next generation military vehicle. Project researchers strongly believe in the benefits that alternative propulsion technologies (such as electric traction, powertrain hybridization, and electrically variable transmission (EVT) concepts) can provide for ground vehicles. However, the authors of this paper feel strongly that design engineers must begin with a clean sheet of paper and capture the system platform as a whole in order to realize the potentials offered by these new technologies. Designers who simply take an existing vehicle and attempt to “swap” powertrain elements are unlikely to substantially improve upon existing “conventional” vehicles. In fact, removing a conventional powertrain from a vehicle which has been specifically developed and refined around a conventional powertrain could potentially lead to an inferior vehicle design. The entire vehicle platform should be designed as an integrated unit (system of systems approach) from ground up, with due consideration being given to the benefits, and potential limitations of the new technologies. Just as one cannot expect to use a “conventional” vehicle with electric powertrain components, one generally cannot bring components from other fields and integrate them into a ground vehicle without realizing any significant modifications. Taking an integrated design approach, each component and subsystem must be based on the overall system

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requirements, as well as the requirements of other sub-systems and components.

DESIGN SOLUTIONS AND IMPLEMENTATION

INITIAL DESIGN REQUIREMENTS

For this specific design application, the vehicle dynamic requirements are dictating the size and the capabilities of the powertrain. Although fuel consumption is a lower priority, it is still very important due to the fact that the quantity of fuel carried for a given mission may exceed 10% of the vehicle's weight. Basics of vehicle dynamics show that this type of vehicle requires roughly 200 km/h (125 mph) on a straight and level improved surface (highway type) –resulting with a power output of 210 kW (282 Hp) on the wheels and over 320 Hp on the prime mover's (diesel engine) shaft. The lion's share of this power (approx 200 Hp) is required to balance the air drag; therefore, special attention should be accorded to the body's shape design –considering drag coefficients on the vehicle's shell. For off-road operation on unimproved surfaces the specific rolling resistance (40-60 Hp on highway) and the suspension dumping may exceed more than 500%; this will require a tremendous amount of power. The minimum power requirements generated by simulations reflect a forecast of 600 Hp continuous and over 1000 Hp for short durations. These short bursts of power are needed for high speed grade operation and/or acceleration. However, for normal traffic operations (approximately 100 km/h - 62.5 mph on highway) the power requirement calculates to about 50-60 Hp. This leads to a minimum power flexibility ratio of 1/10, which exceeds power capabilities of any existing military vehicle known to date. In other words, the powertrain should be able to provide enough power to sustain an Abrams tank at maximum speed and also provide fuel efficient power to a HMMWV operating on the highway. Taking into account the size and weight of available prime movers, the design solution resulted in a dual engine design coupled to an all wheel drive electrically variable transmission (EVT). The AWD-EVT is a ground –up design that leverages the experience gained through OSU in generating new EVT concepts (especially for AWD applications) in past projects and also leverages past experience with the design of high power output electric traction drives. At the time of writing, CAR-OSU holds the electric land speed record with a top speed of 517.8 km/h (321.8 mph). The electric drive used in this application exceeded well over 600 Hp.

DESIGN PROCESS

For all purposes of adhering to a rapid design process and due to the large number of design variables, a single Master Design Point (MDP) strategy was adopted. The MDP is typically identified as the most critical design element and is used as a baseline for the overall design. In this case, a viable crew protective system within this immediate platform class enables our ability to enhance the design to provide an unsurpassed survivability factor. Furthermore, the causality between the MDP and the subsequent design points was enforced as a master-slave relation, further underlining the overall protective strategy and conceptual guiding principle: all vehicle subsystems will be designed devoid of any compromise to crew protection. Subsequently, the vehicle dynamic performances and more importantly the vehicle performances after sustaining different types of threats (mission accomplishment capabilities) were extremely important parameters in the design process –leading to a completely unconventional vehicle topology and powertrain configuration (the concept design's fundamental technical elements and target performance parameters (TPPs) are presented in table 1).

Several other military-specific requirements proved also to have higher precedence than the fuel economy requirement throughout the powertrain design process. In other words, the design process takes anything but a conventional approach to fuel economy.

Requirement	Value	Trend
Crew	4	Up to 6 member or the fifth member horizontally accommodated (a possible wounded)
GVW	4000 – 8000 kg	As light as possible
Wheel base	4000 mm	Should consider as maximum acceptable
Width	2600 mm 102 inch	Should consider as maximum acceptable Should match the Bridge Low
Top speed	200 km/h 125 mph	At least on good roads, objective on sand and/or off road
Step obstacle	400 mm	Should consider as minimum acceptable
Minimum Ground clearance	400 mm	Should consider as minimum acceptable

Table 1: Matrix of Target Performance Parameters

At the time of writing, no known existing military vehicle uses a powertrain based on an EVT configuration, thus a set of design goals were generated in accordance with the vehicle typical usage as follows:

- Operate in limp mode without any electric/electronic device (e.g. batteries, electric machines, inverters, computers etc).
- Full traction on any two wheels (considering the vehicle weight fully supported by any two wheel combination the tractive effort should be limited only by adhesion)
- Gen-set operation power at a sensible level logically equivalent to that of the prime mover power level.

- Ambush maneuvering (tank type cornering)
- Also some other engineering consideration were taken into account based on experience, such as a 7000 rpm max speed constraint on any shaft or gear, use of no lube pumps, and application of modular design concepts.

The EVT basic configuration was chosen based upon the most important vehicle operation which is considered low speed in extreme off road conditions. From a design standpoint this type of operation is also a major input for sizing the EVT and transmission elements. Basically, the maximum torque ratings and load shocks of all the mechanic and electro-mechanic devices (such as gears, bearings, electric machines, CV-joints, and shafts) are dictating more than 80% of the device size and weight. Life and maximum speed have only minor influences in the final size and weight. In general, the input split EVT, currently found in the Toyota Prius design, is considered to be the best option for low speed and urban type driving and potentially could be a candidate for this design. However, an output split configuration proved to mach better for this application, mainly due to the propriety of producing high output torques without the usage of the battery. As it was mentioned prior, the limp mode is favored to provide a higher level of fuel economy. A basic topologic configuration of an output split EVT is shown in fig.1.

The prime mover (diesel engine) and two electric machines (EM1 and EM2) are the driving elements for the EVT. EM1 is coupled to the engine shaft via a gear set. The gear set is also used to match the output characteristics of the two elements. The electric machine is typically faster than a diesel engine. Furthermore, the engine and EM1 are coupled with the EM2 through a differential mechanism, whereas typically one may find using a planetary gear set for this application. The differential mechanism output shaft is driving the vehicle's transmission. This type of output split configuration has two main operation modes. Thus, at low speed EM2 is in generating mode and is counterbalancing the engine torque. The electric energy produced by EM2 is absorbed by EM1, which is taking on a motoring role. In this way the EM1 is increasing the engine torque and an important output torque can be obtained. In practice, the engine torque might be amplified by the EVT several times being equivalent to a classic transmission equipped with a low gear feature. If n is the amplification factor at low speed, thus we obtain n-1 for the maximum tractive effort which can be obtained without the usage of battery;. In other words, the total absence of the battery will diminish the starting traction force with less than 20%. For high speed operation EM2 should change its rotational direction and operate in motoring mode. In this configuration, the EVT output speed could be higher than the engine speed and might be equivalent with an overdrive feature. For high speed operation, the EM1 is functioning in generating mode supplying EM2 with energy. Furthermore, the EVT output is applied to the wheels by a transmission system. A typical transmission system for an AWD vehicle is based on a three differential mechanism topology using a central differential mechanism connected to each axle differential (as shown in fig 1) This type of transmission topology is heavily used world wide for a variety of vehicles and applications despite its major drawbacks. We can conclude that if open differentials are used, then the vehicle's maximum tractive effort cannot possibly reach higher of magnitude than 4 times the worst wheel adhesion condition. In other words, if 3 wheels are located on asphalt and only one on ice, the total tractive effort is not greater than in the case of operation with all the wheels on ice. There are also known sophisticated differential types (such as Torsen, speed-sensitive, self locking, magnetoreological, etc) which are able in some fashion to eliminate or reduce this disadvantage of slippage.

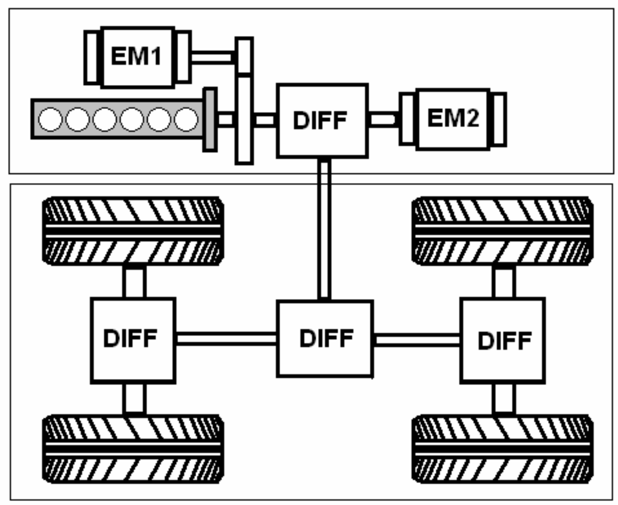


Fig 1: Topology of a classic output split EVT

However, at the time of writing this entire sophisticated differential layout (without exception) is producing significant mechanical losses in situations when functioning in harsh conditions and/or when tight cornering is required. These parasitic mechanical losses at times can reach values as high as 40%. Thus, the authors of this paper strongly believe that for an off-road vehicle, a greater fuel economy may be obtained by developing a new transmission concept, rather than by optimizing the powertrain components. This belief is also based on the obvious fact that a classic configuration (as shown in fig 1) cannot meet the design goals of a new transmission class that was developed with these objectives in mind.

TOPOLOGIC TRANSFORMATIONS

To match the design goals several topologic transformations were created using a step-by-step process.

Step1

The first transformation is based on the fact that there are four differential mechanisms onboard which are used to transfer the power from the engine to the wheels. Fundamentally, one mechanism is fully controlled and the other three are uncontrolled or quasi-uncontrolled. By removing the central differential from the transmission path and locating it in the EVT configuration, only the axles' differentials will remain uncontrolled. The transformation is shown in fig.2. The engine is driving two identically differentials simultaneously and every differential output is driving an axle. The reaction torque on each differential is obtained by splitting the EM2 output in two. This configuration can control the amount of power delivered to each axle with a high level of precision –which is a major advantage in the off-road world. It was also learned that the increased complexity of requiring supplementary power electronics can be traded off with a less complex hardware configuration (e.g. a complex torsen differential can be replaced by a simple planetary gear set). Apparently more complex, the configuration can lead to a superior vehicle behavior and gain more fuel efficiency by principally diminishing the losses in off-road conditions. This configuration is slightly lighter and not necessarily more expensive.

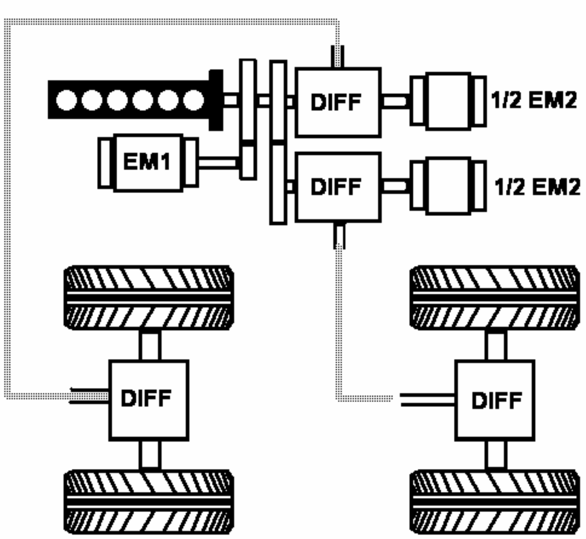


Fig 2: Topologic transformation from classic EVT split

As mentioned earlier in the paper, the new control capability is a major advantage of the new configuration. However, it is also apparent that the new control feature is effective only at low speeds. As an intuitive example for high speed operation, let's suppose a vehicle is traveling at 100 km/h (27.7 m/s or 62.5 mph) on a succession of dry and icy surfaces. There is only roughly a tenth of second in which the driving condition of each axle may differ for each transition (one axle on dry the other on ice), in the rest of the driving conditions for each axle will remain similar. Obviously the new control feature is less effective at high speeds mainly due to the rotational inertias of various components. It is also apparent that there is a higher probability that the lateral condition differs from of the longitudinal condition. As a typical wintertime operating example, a common situation is to travel with the left side wheels on wet asphalt and with the right side wheels on snow.

Step 2

The next topologic transformation was generated to leverage the control features developed by the previous step. Thus, with the same number of elements it is possible to control the amount of power delivered to each vehicle side. The topologic transformation required is shown in fig. 3. Basically, every EVT output is driving one side wheel through a differential mechanism, keeping the same complexity as in fig.2 but introducing some new advantages.

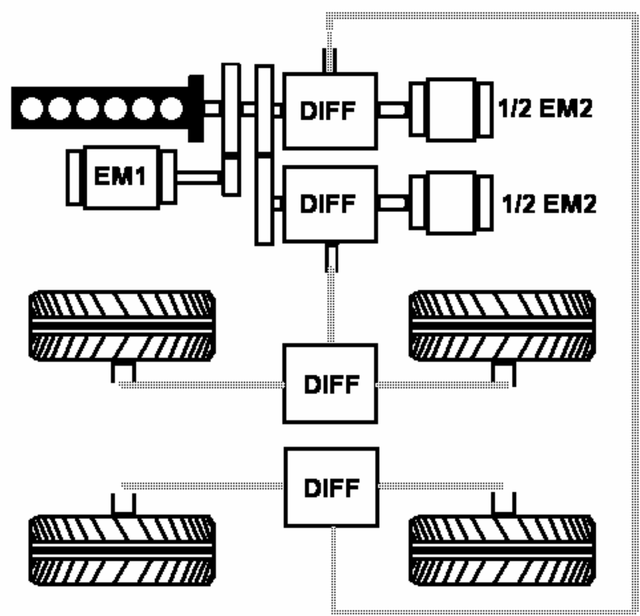


Fig 3: Topologic transformation Step 2

Thus, controlling power delivered side-to-side is an effective and beneficial feature for the entire speed range. Much more, at very low speeds it is possible to reverse the rotational direction of only one of the EVT's output, forcing the vehicle to swing around its vertical axis like a tank. As it was mentioned prior,

implementation of this feature was an important design goal strictly related to the vehicle class. However, the trajectory control which can be implemented by this configuration might be applied as a safety feature for other applications.

Step 3

The next topologic transformation was generated strictly by the vehicle class. It was realized that an all wheel steering vehicle has tremendous advantages in a real combat environment and an all wheel steering system should be one of the top priorities. It was also realized that a symmetric all wheel steering system does not require a differential between the front and the rear wheel. The topologic transformation is shown in fig.4

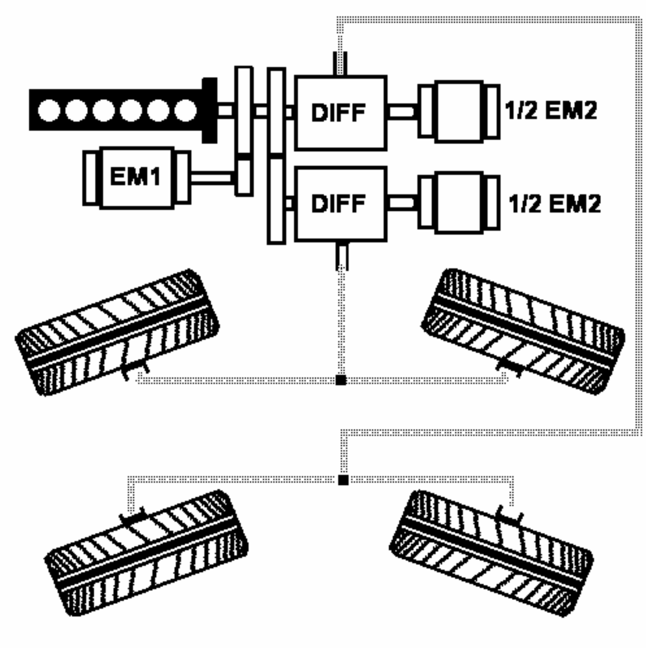


Fig 4: Topological transformation Step 3

When in a cornering maneuver, the same side wheels are steered with the same angle, the wheels' rolling radius is the same, and the wheels' speed will be also be the same; no differential mechanism is needed anymore. For this specific application the CTI system (which is also a must) can guarantee rigorous rolling radius despite different tires wear or loads.

Step 4

If the first three steps are logic synergetic steps, then step 4 represents a fundamentally forward creative step. It consists of the addition of what the authors are baptizing as a "differential shaft." A possible configuration is shown in fig. 5. The differential shaft can be coupled to the two halves of EM2 on a demand basis only. Technically, a gear set is used for each EM2 half and a clutch K is located between the gear sets. Please note that one of the gear sets is changing the rotational sense and the clutch speed (when not engaged) will be roughly double that of EM2. The first benefit of the differential shaft is a parallel mode of operation. Thus, if the clutch is engaged, a reaction torque for both differential mechanisms is produced due to the sense reversal provided by the differential shaft. In this mode of operation, no electric energy flow is needed and the engine power is directly driving the wheels. In most of the cases, this mode is more economic than any EVT mode due to the absence of electric losses and some parasitic mechanical losses. In this mode EM1 is acting in a typically parallel mode, increasing the engine power or producing electric energy. For this specific application the parallel mode is available between 35 km/h and 130 km/h and is considered the main mode of operation. In parallel mode, the differential shaft is also providing the side-to-side mechanical differential action. While performing a cornering maneuver, the differential shaft will rotate slowly in both ways according to what side wheels are spinning faster (left or right cornering). Also in a cornering maneuver, one or both EM 2 can produce or subtract a differential power between the vehicle sides when the differential shaft starts to rotate. In this sense, a higher level of control can be applied for trajectory control and rough terrain negotiation purposes.

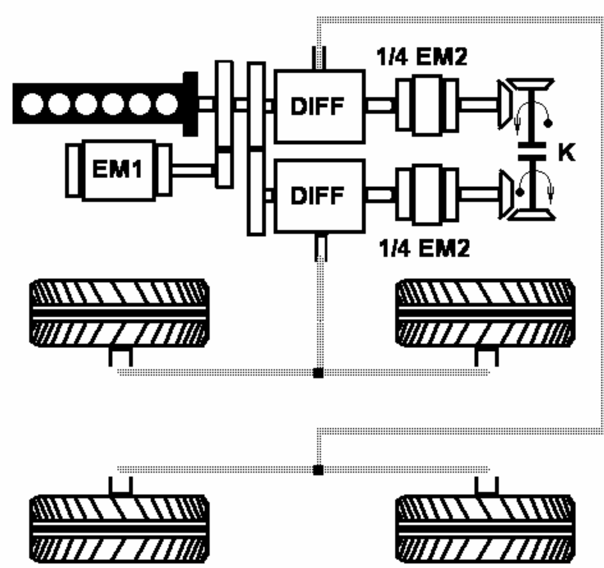


Fig 5: Topological transformation Step 4

The second major benefit of the differential shaft is the reduction of the EM2 size. It was realized that a side-to-side configuration requires the maximum EM2 output capabilities only when the vehicle weight is supported by

only one side of wheels. In a very unlikely event, such as operation at an extreme roll angle, it is possible to obtain a double reaction torque (while traveling at low speeds in EVT mode) by engaging the clutch and reversing (electrically) the sense of one EM2. This feature allows halving EM2 size which is a noticeable gain. Logically, the differential shaft allows the functioning of the powertrain in a very economic parallel mode; it is also providing the hardware for an unprecedented trajectory control. As a result, we realize a savings in weight and overall cost. Using this reasoning, the authors are considering the differential shaft the most important achievement of the concept design.

Step 5

As it was mentioned prior that the engine power requirement exceeds 600 Hp, which drastically narrows the commercially available options. It was also apparent that the classes of newly developed high speed diesel engines are providing a much better power density than that of a typical heavy duty truck type diesel. However, at the time of writing nothing was available in the needed power range. Thus, a dual engine solution was adopted. Fig. 6 shows the topology modified to accommodate the two diesel engines. For all practical purposes, the EVT was split into two –one for each side of the vehicle. The diesel shafts are geared together by using EM1 as a transmission shaft. Each diesel is equipped with a shaft clutch (not shown) allowing the system to operate with a single engine, both engines, or no engine at all (silent mode or regenerative mode) dependent on what is the most economic functioning configuration during various conditions (i.e. variable speeds and terrains). Despite the fact that the fuel saving feature is some what important, the geometrical distribution of the powertrain components is providing the most important design feature of all – the crew protection. Basically, it is possible to locate the crew compartment (as an island) in the middle of the EVT's massive elements and to use the EVT as supplementary protection (ballistic and anti-intrusion).

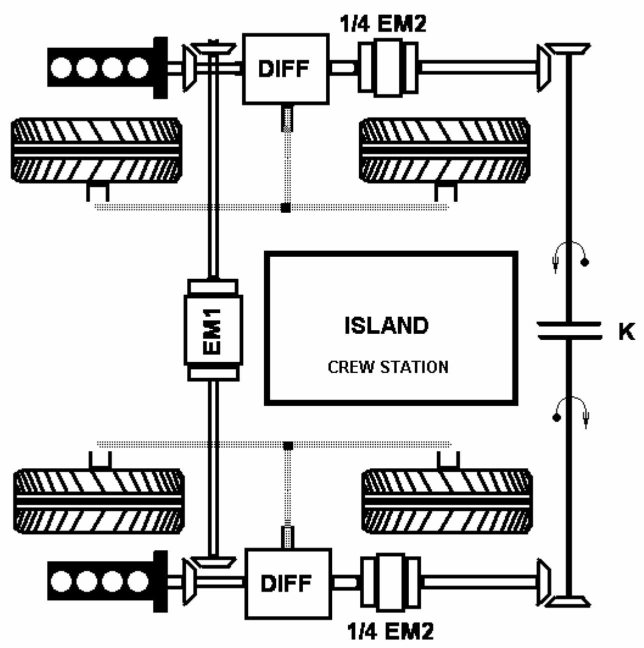


Fig 6: Two diesel EVT topology (Step 5)

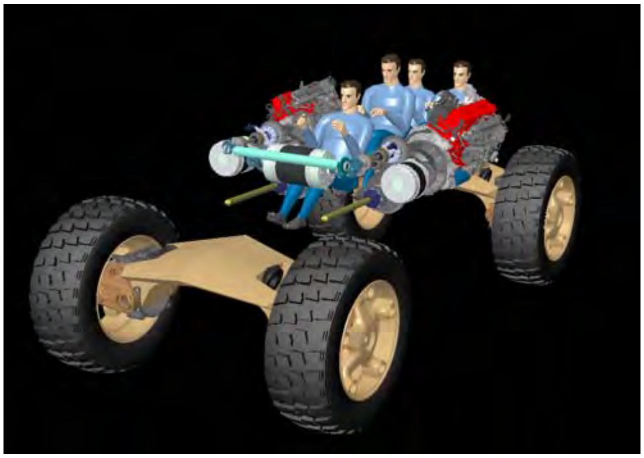


Fig 7: 3-D Rendering of crew and NGV – ISLAND CONCEPT of Crew Protection System

Step 6

Obviously, the configuration shown in fig. 6 is providing a high number of limp mode capabilities for it can function only with half of the elements. This configuration meets all of the design requirements. However, in order to meet the main limp mode requirement (functioning without any electric/electronic device) another transformation was needed. As it was mentioned prior, the parallel mode is available anywhere between 35 and 130 km/h and the limp mode is equivalent to a parallel mode in which EM1 is not available. For practicality sake, the parallel mode range is possible to match the limp mode requirement by extending down to zero speed. Fig. 8 shows the final configuration. Thus, the clutch K function was extended by replacing the regular clutch with two multi-disk clutches (KM) each located directly in the EVT's bodies.

Kinematically, each clutch is placed in front of the differential shaft (which, in this configuration, is a real full shaft) avoiding some mechanical losses in EVT mode, but mainly halving the clutches max speed operation. The multi-disk clutches are rated to sustain a 10 launch sequence without exceeding the heat dissipation capabilities. Furthermore, a very simple mechanical system was designed in parallel with the complex control system, which allows the driver to engage the clutches with a conventional pedal.

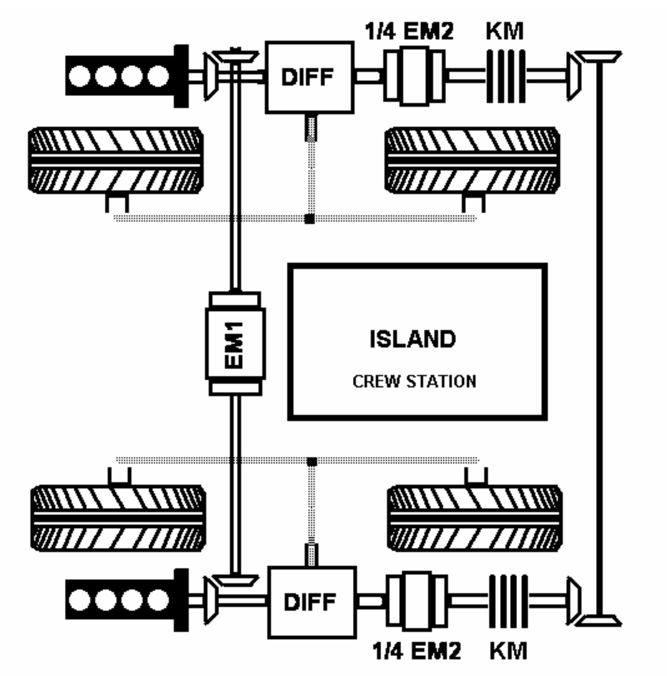


Fig 8: Final EVT Concept design (Step 6)

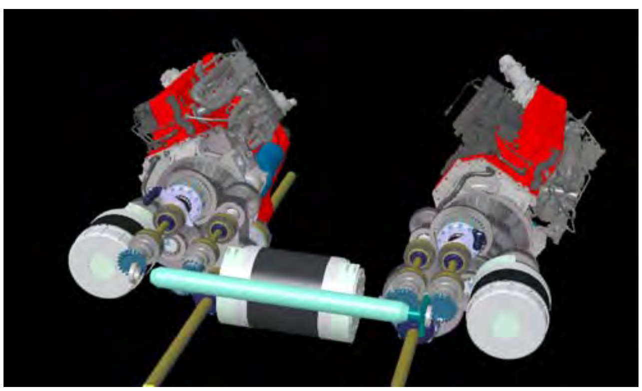


Fig 9: 3-D Rendering of ISLAND CONCEPT EVT

The EVT architecture is based on two units formed by existing Cummins engines (350 Hp each) and a corresponding gear box located on each side of the vehicle. The diesel units require only minor modifications to the oil pan and clutch bearings. The gearboxes are connected through the central electric machine EM1 located in front of the driver. There are 4 output shafts – one for each wheel. The gearbox architecture was designed to utilize the least amount of components, yet provide true combat capabilities and functionality during operation. Although the gearbox layout and operation appears to be complex, its overall complexity is equivalent (or below) to that of a conventional transmission. Figure 10 shows a cutaway view of the gear box.

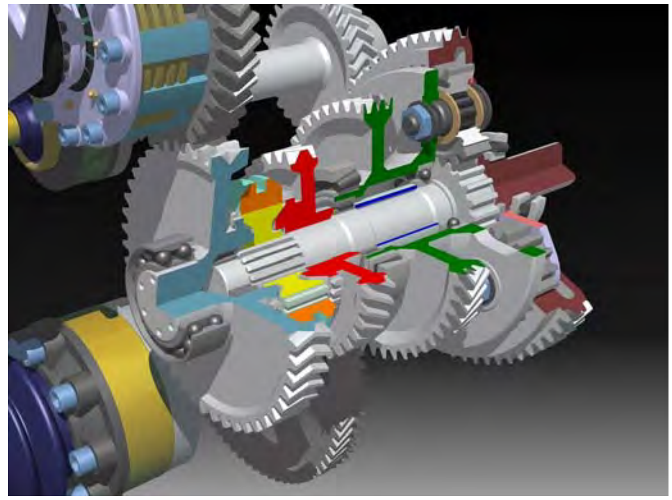


Fig 10: Cutaway view of gearing

The planetary gear set can be represented as in figure.11 The kinematic equation that relates the speeds and the torques on the various components can be expressed as follows:

$$\begin{cases} \dfrac{-T_c}{1+\gamma} = \dfrac{T_r}{\gamma} = T_s \\ \gamma \cdot \omega_r + \omega_s = (1+\gamma)\omega_c \end{cases}$$

Where *T* is the torque, ω is the speed, the subscript *r* refers to the ring, *s* to the sun, and *c* the carrier. The coefficient γ is the planetary gear ratio, defined as the ratio between the number of teeth of the ring and the sun:

$$\gamma = \frac{n_r}{n_s} = 2.5\,.$$

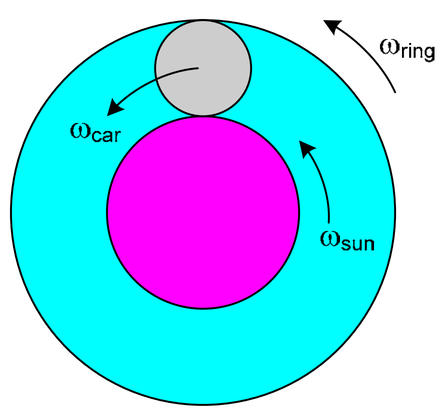


Fig 11: EVT model – Planetary Gear Set

DYNAMIC SIMULATION

The dynamic simulator built for the vehicle consists of 4 major components:

1. Drive cycle + driver block, which generates the velocity profile and controls the vehicle speed;
2. Vehicle dynamics model, which calculates the vehicle velocity given the torque generated by the powertrain;
3. Powertrain model, where all the powertrain components (engines, electric machines, transmission elements) are connected together and the tractive force/torque is calculated using the driver input
4. Energy management system, which translates the driver commands into torque request for the engine and/or the electric machines, using a control strategy that minimizes the fuel consumption.

DRIVE CYCLE + DRIVER BLOCK

The drive cycles are loaded as time histories of required vehicle speed. The driver subsystem is essentially a PID controller. Basically, the speed of the vehicle is fed back into the driver block and compared to the desired vehicle speed as determined by the chosen driving cycle. The PID controller then minimizes the error between these two values. From the desired vehicle speed, the subsystem computes an accelerator pedal position, alpha, and a brake pedal position, beta. These two values are sent to the powertrain and converted by the controller into a torque command for the internal combustion engine and the brakes when necessary.

VEHICLE DYNAMICS MODEL

The vehicle dynamics model determines the vehicle acceleration given the torque generated by the powertrain and the resistant torque due to the rolling / aerodynamic resistance:

$$a = \frac{1}{m_{eq}}\left(F_{trac} - F_{rr} - F_{aer}\right)$$

where m_{eq} is the equivalent vehicle mass (sum of the actual mass and the equivalent inertia of the rotating elements), F_{trac} is the tractive effort generated by the transmission, F_{rr} is the rolling resistance, F_{aer} is the aerodynamic resistance.

POWERTRAIN MODEL

In the configuration examined, the vehicle is a parallel hybrid. The electric motor and the engines are rigidly connected (tightly coupled) and the torque entering the planetary device is the sum of their torque outputs. The power request can be satisfied from either the internal combustion engine, or the electric machine, or a combination of both. The dynamic model is built using efficiency maps for engines and electric machines, and taking into account their inertia (but neglecting the inertia of the gears).

The configuration *described* can be represented as in the following figure:

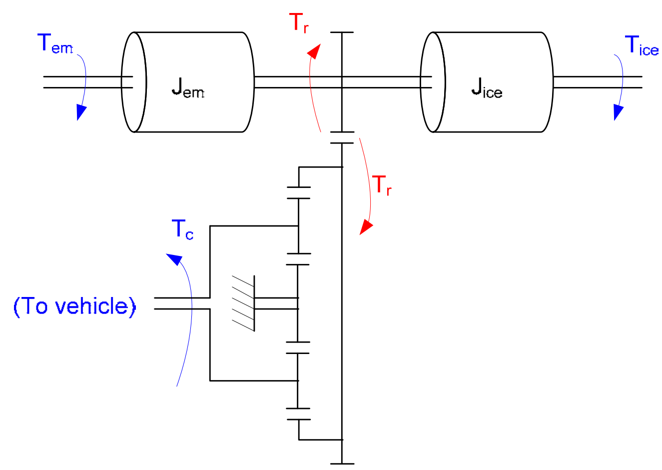


Fig 12: Powertrain Configuration in Parallel Mode

It is basically a fixed-ratio transmission between the prime movers and the wheels, which are connected to the carrier via another fixed gearing.

To describe the transmission of torque and speed in this case, the equations of the planetary gear set are used in the following form:

$$
\begin{cases}
T_c = \dfrac{(1+\gamma)}{\gamma} T_r \\
T_s = -\dfrac{1}{\gamma} T_r \\
\omega_r = \dfrac{(1+\gamma)}{\gamma}\omega_c - \dfrac{\omega_s}{\gamma}
\end{cases}
$$

In other words, the torque generated by the engine and the electric machine is the input to the vehicle model, and the engine/electric machine speed is depending on the vehicle speed.

ENERGY MANAGEMENT SYSTEM

The powertrain described can be used in several different configurations and offers a great degree of flexibility. However, for a first cut, the hybrid strategy included in this model is a simple "thermostatic" controller of the battery state of charge: the electric motor is used as much as possible to generate torque if the SOC of the battery is above the minimum value (set to 55%), and to recharge the battery as long as the SOC is below the maximum value (80%). When the torque delivered or absorbed by the electric machine is not adequate enough to satisfy the driver's demand, the internal combustion engine delivers the remaining power.

SIMULATION RESULTS

The hybrid configuration has been tested on a driving cycle representative of extra-urban driving conditions, with speed variable between 20 and 75 mph (see figure 13)

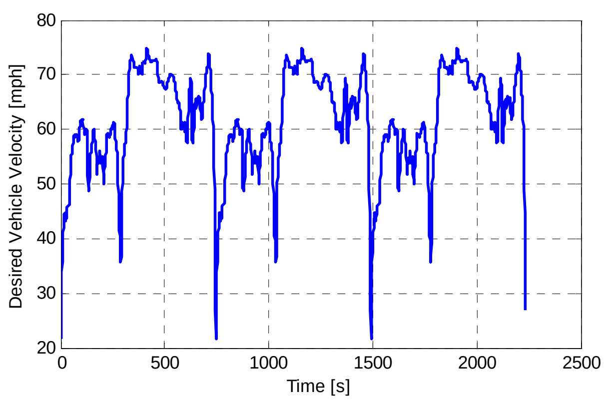
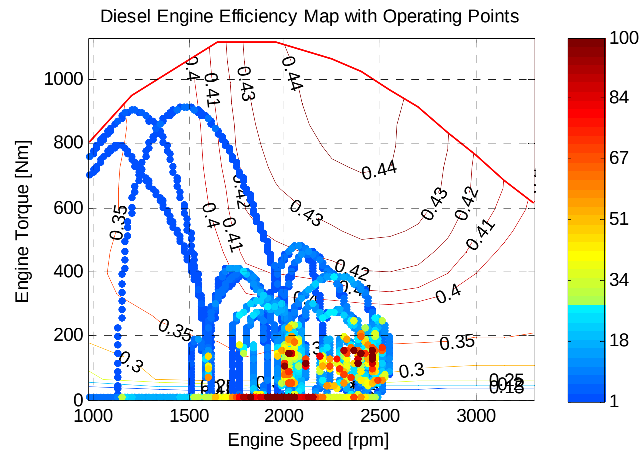


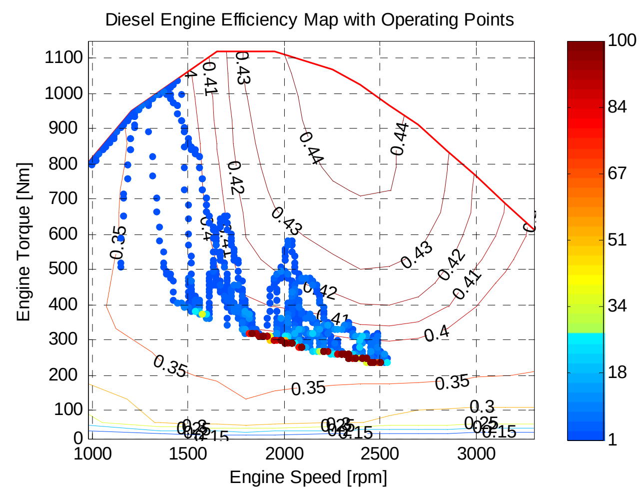
Fig 13: Test result in extra-urban driving conditions

The results in terms of fuel consumption for different cases are listed below:

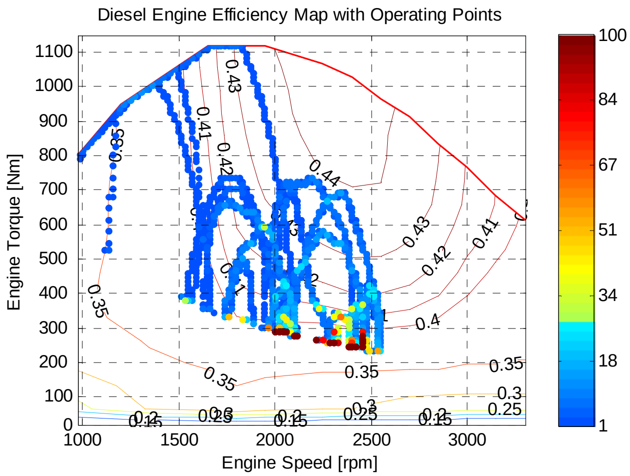
a) 2 engines running, no batteries: 8.8 mpg

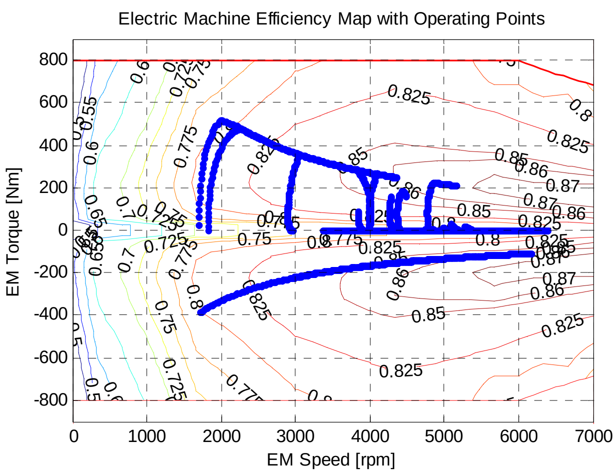


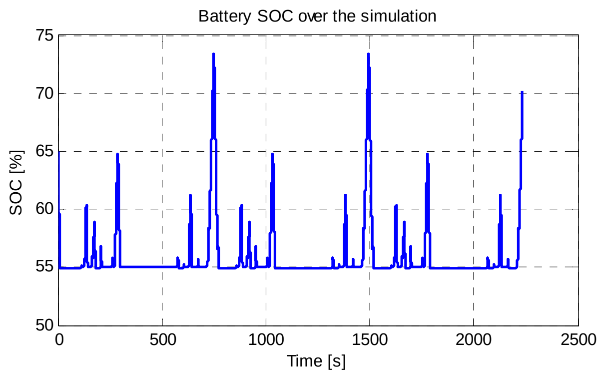
b) 1 engine running, 1 off, no batteries: 10.0 mpg



c) 1 engine running, batteries on (parallel hybrid): 11.3 mpg







CONCLUSION

Overall, the design for the EVT concept has been progressing at a phenomenal rate and we have been able to meet several platform design requirements such as enhanced survivability, dynamic performance, and mobility. As we develop the Next Generation Military Vehicle (NGV) further, we gain a better understanding of how this subsystem feeds into the overall platform and is able to meet the functional needs.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

AWD: All Wheel Drive

AWS: All wheel steering

CTI; Central Tires Inflation

EVT: Electrically Variable Transmission